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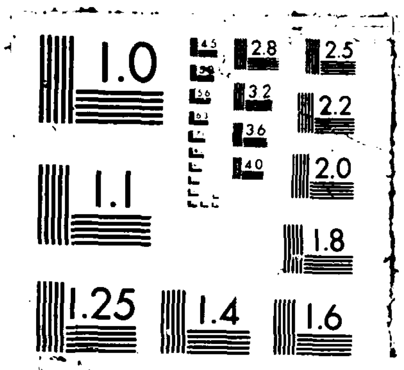
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HIGH REYNOLDS NUMBER PUMP FACILITY FOR CAVITATION RESEARCH

by

K. J. Farrell, M. W. McBride, M. L. Billet

ARL-PSU

TECHNICAL REPORT

The Pennsylvania State University
APPLIED RESEARCH LABORATORY
P.O. Box 30
State College, PA 16804

HIGH REYNOLDS NUMBER PUMP FACILITY FOR CAVITATION RESEARCH

by

K. J. Farrell, M. W. McBride, M. L. Billet

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A High Reynolds Number Pump Facility (HIREP) designed for cavitation studies in the blade-tip/end-wall region of an axial flow pump is described in this paper. The facility consists of a 1.07-m diameter pump state driven by a 1.22-m diameter downstream turbine. An incompressible Reynolds Number of 6×10^6 at the rotor tip is achievable. The two units rotate on a common shaft and operate in the 1.22-m diameter test section of the Garfield Thomas Water Tunnel of the Applied Research Laboratory Penn State. The facility was designed to accomodate laser velocimeter (LV) measurements in the pump stage, radially traversing five-hole probes in every stage, and a number of transducers in the rotating fram of reference: steady and unsteady pressure transducers force and torque cells, and accelerometers. The latter capability is provided by a slip-ring unit and hollow blade passage ways for conductors from the instrumentation in rotor-tip region. An optical quality window for LV measurements and other windows and ports are available for visual observation and instrumentation access.

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ABSTRACT

A High Reynolds Number Pump Facility (HIREP) designed for cavitation studies in the blade-tip/end-wall region of an axial flow pump is described in this paper. The facility consists of a 1.07-m diameter pump stage driven by a 1.22-m diameter downstream turbine. An incompressible Reynolds Number of 6×10^6 at the rotor tip is achievable. The two units rotate on a common shaft and operate in the 1.22-m diameter test section of the Garfield Thomas Water Tunnel of the Applied Research Laboratory Penn State. The facility was designed to accomodate laser velocimeter (LV) measurements in the pump stage, radially traversing five-hole probes in every stage, and a number of transducers in the rotating frame of reference: steady and unsteady pressure transducers, force and torque cells, and accelerometers. The latter capability is provided by a slip-ring unit and hollow blade passage ways for conductors from the instrumentation in the rotor-tip region. An optical quality window for LV measurements and other windows and ports are available for visual observation and instrumentation access. Advanced instrumentation systems for blade static pressure measurement, rotor-tip/end-wall gap measurement, and cavitation viewing were developed. The hydrodynamic and mechanical design, the instrumentation system, and the operating characteristics of HIREP are discussed.

NOMENCLATURE

A	flow annulus area - 0.8938 m^2
C_Q	dimensionless torque coefficient - $Q/(\rho V_{ref}^2)AD/2)$
D	pump diameter - 1.0668 m
n	rotational speed (rev/sec)
P_d	pressure of desinent cavitation
P_o	stagnation pressure
P_v	vapor pressure
PR	ratio of shaft power to fluid power - $Q\omega/(\rho V_{ref}^3 A/2)$
Q	shaft torque
U	tip speed
V_{ref}	axial velocity at inlet
Δ	change in any quantity
η	pump rotor efficiency - $\Delta P_o V_{ref}\ A/Q\omega$
ξ	turbine inlet guide vane pitch angle
ρ	fluid density
ϕ	dimensionless flow coefficient - $V_{ref}\ A/nD^3$
c	dimensionless head coefficient - $\Delta P_o/\rho n^2 D^2$
σ	cavitation number - $(P_d - P_v)/(\rho U^2/2)$
ω	rotational speed (rad/sec)

INTRODUCTION

Axial flow pumps are subject to many of the the same design considerations as axial flow compressors with the important exceptions that compressibility effects are absent and cavitation can occur. Multi-blade row pumps are being designed with either inlet guide vanes or downstream stators. It is important to understand the three-dimensional flows that occur in multi-blade row pumps since departures from design conditions lead to excessive incidence variations and hence cavitation. The tip clearance region of the rotor accounts for a substantial portion of the energy loss in a pump and is a source of cavitation. In the blade-tip/end-wall region, the flow does not obtain the theoretical momentum increase, and the interaction of this tip clearance flow with the incoming secondary flows, blade boundary layer, and annulus-wall boundary layer disturbs the through-flow in a significant portion of the blade passage.

Extensive research has been completed in the axial flow compressor field on tip clearance flow [1,2]. Lakshminarayana [3,4] has developed several models of the leakage and secondary flows. The work of Rains [5] represents a major attempt to correlate a model of clearance flows with cavitation in pumps.

At this time, no complete model of the complex flow in the tip region of a pump exists. A numerical solution of this complex flow via the Navier-Stokes equation is not forthcoming due to the existence of a complex three-dimensional turbulent flow and to the lack of a good experimental database by which to model the flow. A limited experimental database does exist [5,6,7,8]; however, details of the tip flow were not obtained in any experiment due to the small size of the pumps tested. A large pump is required to characterize the flow phenomenon in this region.

The High Reynolds Number Pump (HIREP) facility has been designed, fabricated, and tested at the Applied Research Laboratory. The purpose of this facility is to provide an axial-flow multi-blade row pump of sufficient size to accomodate a variety of instrumentation in both a stationary and rotating frame. A description of the hydrodynamic and mechanical design, the instrumentation system, and the operating characteristics of the facility is presented.

HYDRODYNAMIC AND MECHANICAL DESIGN

A mechanical layout of the HIREP facility is shown by Figure 1. The relationship of the HIREP facility to the overall Garfield Thomas Water Tunnel facility is shown by Figure 2. The HIREP facility is a turbomachine consisting of two basic units coupled by a common shaft. The upstream portion is the primary, instrumented axial flow pump stage. The downstream portion is a regenerative turbine. The main drive motor of the Water Tunnel provides sufficient energy to overcome the losses incurred in the pump and turbine stages and throughout the loop. The facility is analogous to the gas turbine engine where the air is compressed, heated in the combustor, and expanded through the turbine. The energy that would be supplied by a combustor is supplied by the tunnel main drive motor. A total of five blade rows comprise the HIREP facility.

The upstream portion of the pump stage consists of a set of inlet guide vanes which provide a specified distribution of swirl at the rotor leading edge and generate viscous wakes and secondary flows typical of axial flow pumps. These wakes and secondary flows interact with the rotor and have substantial influence on its performance, especially with regard to cavitation onset and type, and the generation of unsteady pressures and forces. The inlet guide vanes also provide structural support for the forward centerbody. The flow entering the unit from the water tunnel nozzle is accelerated through the annulus formed by a liner and forward centerbody and is essentially uniform in the circumferential direction. The inlet guide vane set consists of a number of individual blades with mounting blocks so that the hydrodynamic design of the vanes may be changed and retrofit with relative ease.

The second blade row in the system is the primary pump rotor. The rotor consists of a hub with individual blades attached to mounting blocks. The hydrodynamic shape of the blades is determined through consideration of spanwise loading, the amount of swirl generated by the inlet guide vanes, and the power available from the drive turbine. The rotor-tip/end-wall gap may be adjusted on each blade by changing the thickness of spacers installed under the blade mounting blocks. Several blade designs currently exist, each having some parametric design variation included. In addition, some individual blades have provision for special instrumentation which will be discussed subsequently. A spanswise cavity in the blade provides a passage way for wire leads to pass from the blade-tip to the chamber in the hub. The hub is bolted to a large disc which is welded to the upstream end of the driveshaft. The forward facing end of the hub is open to allow access to the instrumentation and is sealed with a plate after electronic connections have been made. The baseline rotor design is capable of absorbing 0.97 MW at a flow velocity of 15 m/s through the pump. This condition has a rotor blade chord Reynolds number of six million based on the relative velocity. This is roughly six times the maximum value attainable with model scale pumps currently in use for research purposes.

Immediately downstream of the rotor is a set of struts whose primary purpose is support of a main shaft bearing. The strut hub also supports the upstream end of the center body housing which contains lubrication for the bearing and provides an attachment for the turbine inlet guide vane pitch mechanism. The hydrodynamic design and location of these struts provides a means to measure unsteady pressures associated with rotor blade wakes, simulating the presence of a stationary blade row. Furthermore, the struts could easily be replaced by a set of downstream stators designed for the pump rotor.

The fourth blade row is the set of adjustable turbine inlet guide vanes. The pitch angle of these vanes is continuously variable from minus forty degrees to plus fifteen degrees, relative to the shaft centerline. This variability allows the system to operate from nearly zero power to one hundred fifty percent design power. The adjustment capability is external to the facility and thus allows variability during operation. A swash plate forms a common slider for fourteen slider-crank linkages, one for each vane, providing simultaneous adjustment without binding. The plate is keyed to a centerbody support which surrounds the drive shaft and is also aligned by two linear bearings which travel on guide bars. A power screw moves the plate axially.

A set of miter gears on the upstream end of the power screw transmits control to a handwheel exterior to the pump via a shaft housed in the support strut immediately aft of the pump rotor. An indexing device on the crank mechanism indicates the pitch of the turbine inlet guides vanes. The large mechanical advantage of the mechanism allows easy manual adjustment during pump operation at axial velocities of up to 8 m/s. A locking mechanism prevents any movement during operation.

The fifth blade row is the regenerative power turbine. The blading in this row has the capability of incremental variable pitch, in order to match a test rotor different from the baseline design. The blades are integrally cast with a cylindrical base which can be rotated in the hub to obtain pitch adjustment. Once the pitch is set, the blade is held stationary by a set of bolted segments which together form a retaining ring. Repeatablity of position is maintained by a guide pin which has four discrete positions, each 2.5 degrees apart. The downstream end of the shaft is supported by a cruciform strut assembly which houses a thrust bearing for the drive shaft. Another strut further downstream forms a vertical, hydrodynamically faired enclosure for the instrumentation leads connecting the stationary end of the slip-ring unit with the data rack and power supplies.

The hydrodynamic design and flow analysis of the HIREP facility was performed utilizing the Streamline Curvature analysis method of Reference [9]. The method is an Euler solver which includes loss models associated with each blade row. The results of the analysis are predictions of velocity and pressure profiles at blade row leading and trailing edges. The input to the analysis consists of spanwise and chordwise loading distributions for each blade row. In the case of the HIREP facility, four blade rows are solved simultaneously, with the loading and losses of upstream rows affecting the design of downstream rows. The torque on the rotor and turbine are balanced by way of modifying the spanwise loading on the turbine. The analysis predicted an overall hydraulic efficiency of seventy-five percent, relative to generated power on the HIREP drive shaft. The concept of a regenerative turbine drive for a pump rotor led to questions of risk, especially with regard to system starting and stability. To verify the concept,a one-sixth scale model of the HIREP facility was fabricated and tested with air as the working fluid. A photograph of the air model is seen in Figure 3. A bell-mouth inlet was installed on the upstream end while a transition into a centrifugal blower was installed downstream of the turbine. Using a variable speed blower drive, it was possible to achieve velocities of 15 m/s in the model HIREP working section. The rotational speed corresponding to this velocity was 1680 RPM, at design. No undesirable transients or unstable operating characteristics were noted during the testing of this model. On the basis of these results and of the system analysis described above, the full size HIREP facility was mechanically designed and fabricated.

INSTRUMENTATION SYSTEM

The facility contains provisions for a wide range of fluid dynamic measurements and methods. The 120-channel, low noise slip-ring unit accomodates many measurements in the rotating frame with ease comparable to that of a stationary measurement. The large, hollow chamber in the rotor hub houses an

assembly of insulated displacement connectors which form the termination of the leads exiting the rotating end of the slip ring unit as shown in Figure 4. From this termination, four ribbon cables with high flexing capability carry signals to the rotating end of the slip-ring unit through the inner diameter of the drive shaft. Ribbon cables form the connection between the stationary end of the slip-ring unit and the data rack and power sources outside of the facility. The signals from the transducers are sampled and processed through a data analysis and reduction system consisting of the driving software program, a low pass filter, an integrating voltmeter, a multiplexer, and a VAX computer system. The rotor blades on which the tip clearance flow and cavitation studies have been performed contain a hollow cavity through approximately 80% of the span from hub to tip. The cavity provides a passage for conductors from transducers in the rotor-tip to reach the mass termination in the hub chamber. The passage is filled with foam rubber to prevent chaffing from cable flexing due to centrifugal force. Clips attached to the chamber wall hold the cable from the base of the rotor blade to the connector assembly described above. The facility is housed in the 1.22-m diameter test section of the Garfield Thomas Water Tunnel, where numerous windows and ports are available for visual observation and instrumentation access.

Velocity and Pressure Measurement

Five luminaire windows on the top cover of the test section were replaced with drilled metal inserts to attach traversing five-hole probes. The probes were located at the inlet and exit of the rotor, at the inlet of the turbine inlet guide vanes, at the exit of the turbine inlet guide vanes, and at the turbine exit. A pitot-static probe was placed upstream of the rotor inlet guide vanes where the centerbody and liner are parallel and was used to determine the axial velocity through the pump. Two small holes were drilled in the top cover of the test section for the pitot-static probe and a static pressure port. A total pressure probe in the nozzle upstream of the test section and the static pressure port were also used to determine the axial velocity through the pump. This measurement was used as the reference axial velocity and for data normalization. The tunnel pressure control system was connected to the static pressure port. The hydraulic lines were connected to a bank of differential pressure transducers located at a height equal to that of the water tunnel centerline. This method eliminates the need to calculate the static pressure head between the various probes. The standard viewing window adjacent to the rotor was replaced by an optical window suitable for LV measurements of the rotor inlet, exit, tip-gap, and passage flows together with the shaft encoder signal.

Force and Torque Measurement

Forces and torque are measured on an individual rotor blade and the drive shaft. The rotor blade force is measured by two two-component shear beams which are mounted in the base and extend radially outward to support the lower portion of the blade span. The axial and tangential forces and the pitching moment can be derived from these measurements of strain. On another blade, the outer ten percent of the blade span is instrumented with a two-component force balance to measure the normal and chordwise forces. The inner diameter of the drive shaft is strain gauged to measure torque. The signals and powering leads are connected

to a connector plate in the hub chamber. The excitation voltages for the strain gauges are remotely sensed to correct for line loss.

Blade Static Pressure Measurement

Several static pressure transducers are located in the tip region of the rotor blade and in the gap. The transducers consist of semiconductor strain gauges which are bonded to a stainless steel diaphragm. The leads from the transducers are channeled through milled slots on the blade surface to a center hole which is drilled through the pressure and suction surfaces to a center cavity. The leads enter the cavity at this hole and are soldered to a small printed circuit board. The board contains a right-angle header connector so that the ribbon cable lying in the hollow passage of the rotor span can easily be connected to this socket and the tip restored to its correct position. Epoxy resin is used to restore the continuity of the blade surface and to seal the holes where the tranducer leads enter the center cavity. The transducers were mounted beneath a Helmholtz cavity to protect the diaphragm from high, localized pressures caused by the collapse of cavitation bubbles. During the assembly of the instrumented blade, petroleum jelly was placed into the cavity so that when the transducer was put in place, it entirely filled the space, and no trapped air was present. The acoustic impedance of petroleum jelly closely matches that of water for unsteady pressure measurements. A transducer was also mounted on the end-wall in the same manner.

Dynamic Rotor-tip/end-wall Gap Measurement

A dynamic measuring system was developed to measure the rotor-tip/endwall gap during HIREP facility operation. A variable impedance transducer was placed in the liner surrounding the pump such that the face of the transducer was flush with the inner diameter of the liner. The impedance variation is caused by the occurrence of eddy currents in the conductive, metallic rotor-tip. The coupling between the coil in the sensor and the rotor-tip is dependent upon their relative displacement. The gap of any particular blade can be measured by sampling the signal from the transducer during the appropriate angular positions of the rotor as triggered by a conditioned signal from the shaft encoder. The resulting data are scanned for the minimum voltage which occurs when the blade-tip is directly over the top of the sensor face. The system consisted of a fifteen volt power supply, an oscillator, linearization network, amplifiers, a demodulator, and an A/D data acquisition system.

Cavitation Viewing

A low-RPM video viewing system was developed to view cavitation on the facility. At high rotational speeds, the blade appears stationary as viewed by the eye due to synchronous stroboscopic lighting. At slower rotational speeds, like those of the HIREP facility, the eye is unable to integrate the images to form a continuous image. The low-RPM video viewing system captures a stroboscopically illuminated image and maintains a display of this image until the next image has been digitized and placed in memory. Cavitation tests can be documented on a video cassette for post-test review.

Angular Position Measurement

An incremental optical encoder on the downstream end of the shaft was used for several measurements. The incremental output of the encoder was conditioned in the encoder interface unit to provide a ramp waveform which peaks once per revolution. The RPM was measured by counting the home pulses of the encoder. The encoder interface unit also produces a trigger signal for the strobe lighting system and the low-RPM video viewing system. A second unit was used to provide blade position to the rotor-tip/end-wall gap measuring system.

OPERATING CHARACTERISTICS

The evaluation of the facility's performance was completed by measuring shaft torque and fluid total pressure at each station as a function of velocity and flow coefficient. The flow coefficient was varied by changing the pitch angle of the turbine inlet guide vanes. The axial velocity through the pump was changed by adjusting the RPM of the water tunnel main drive motor. The velocity was first set at 4.6 m/s and then increased in four increments. At each axial velocity, four flow coefficients were tested. The radial location of the five-hole probes was the theoretical fifty percent mass flow streamline as determined by the Streamline Curvature solution of the through-flow. The probes provided a measurement of total and static pressure as well as the velocity magnitude and direction. The limitations of single point measurements, when used to quantify the performance, should be noted. The instrumentation block diagram for this performance test is noted in Figure 5. The shaft of HIREP begins to turn at an axial velocity of approximately 1.5 m/s and rotates at approximately 40 RPM. Axial velocities of up to 15 m/s are possible with rotational speeds reaching approximately 400 RPM. The operating static pressure range at the inlet to the machine is 42 to 345 kPa. Measured values of the performance parameters are compared to the design values in Table 1.

The flow coefficient as determined by the reference velocity and the rotational speed is shown in Figure 6 as a function of turbine inlet guide vane pitch angle. The flow coefficient was found to decrease with increasing axial velocity through the pump. The flow coefficient also increased with increasing turbine inlet guide vane pitch. The increase in pitch reduced the amount of swirl into the turbine; consequently, the turbomachine operated at at a lower torque and RPM.

The torque on the drive shaft which couples the pump and turbine is shown in Figure 7 as a function of flow coefficient. At constant rotational speed, the torque is equal to the pump rotor torque and the turbine rotor torque. The torque coefficient was found to decrease with increasing axial velocity through the pump. A similar Reynolds number dependency was noted in Figure 6. Changes in axial velocities at higher velocity values have a lesser effect on the torque coefficient.

Two five-hole probes, one located 4 cm axially upstream of the rotor root leading edge and the other 5 cm axially downstream of the root trailing edge, were used to determine the total pressure rise over the rotor at the theoretical fifty percent mass flow streamline as a function of the flow coefficient.

The dimensionless rotor head rise coefficient is shown in Figure 8 as a function of flow coefficient. For a particular flow coefficient, the pressure rise reduces with increasing relative velocity and RPM.

The total pressure drop over the turbine was measured in a similar fashion. The probes were located 10 cm axially upstream of the turbine root leading edge and 5 cm axially downstream of the turbine root trailing edge. The dimensionless head drop coefficient at the fifty percent mass flow streamline is shown in Figure 9 as function of flow coefficient. The head coefficient is maximum and minimum at flow coefficients of around 1.33 and 1.42, respectively. At high flow coefficients, the turbine inlet flow has minimal swirl because the inlet guide vanes are nearly axial in position. Consequently, the pressure drop across the turbine is reduced. At low flow coefficients, the fluid separates from the turning vanes, and again the effectiveness of the turbine is reduced.

The pump rotor efficiency was defined by the product of the flow rate and the total pressure rise over the pump rotor divided by the shaft power. This efficiency is shown in Figure 10 as a function of flow coefficient. At constant flow coefficient, the efficiency reduced as the axial velocity was increased. An efficiency-like parameter, the power ratio is shown in Figure 11. This quantity is the ratio of the shaft power to the total fluid power. As the axial velocity increases, this quantity decreases because a larger portion of the fluid power must be provided by the water tunnel main drive.

Cavitation Testing

A typical photograph leakage vortex cavitation is shown in Figure 12. The vortex is formed by the interaction of the tip clearance flow with the through-flow. The location of the vortex along the blade chord was observed to vary with gap spacing for a fixed tip geometry. For very small gaps, the vortex was observed to eminate near the blade leading edge, and for large gaps the vortex moved to the trailing edge.

The cavitation number was found to depend not only on the rotor-tip/end-wall spacing, but also on the relative velocity for a fixed tip geometry. Figure 13 illustrates the dependency of the cavitation number on the rotor-tip/end-wall gap and the relative velocity. Previous results [6,8] have shown this relationship; however, the differences in the shape of the curve are difficult to explain at these high local Reynolds numbers of approximately six million.

SUMMARY

A large, axial flow pump has been constructed at ARL/PSU in order to provide a research facility for the study of flow at blade chord Reynolds numbers of six million, particularly in the rotor-tip/end-wall region. This area of a turbomachinery flow field is a source of efficiency loss and cavitation. The axial flow facility consists of a pump rotor and a regenerative power turbine coupled together and contained in the 1.22-m diameter test section of the Garfield Thomas Water Tunnel. Both stages have a set of inlet guide vanes, and three structural vanes are immediately downstream of the pump rotor. The

flow coefficient is variable from 1.25 to 1.45 by adjusting the pitch of the turbine inlet guide vanes, which are continuously variable, and by adjusting the pitch of the turbine blades, which are discretely variable. The operation of the pump/turbine was remarkably stable from 1.5 to 15 m/s with rotational speeds of 40 to 400 RPM. This range corresponds to a blade chord Reynolds number of one-half million to six million. The large size of the HIREP facility accomodates a variety of instrumentation in both a stationary and rotating reference frame. One hundred twenty instrumentation leads are available on the rotating reference frame alone. Furthermore, several advanced instrumentation systems were employed with success: the blade static pressure measurement, the dynamic rotor-tip/end-wall gap measurement, and the low-RPM viewing system for cavitation calling. An optical window over the pump rotor stage allows laser velocity measurements of the rotor inlet, exit, passage, and tip-gap flows. The capability to achieve high Reynolds number flow and to accomodate a variety of instrumentation render the HIREP facility a unique and invaluable tool for fluid dynamicists to understand the physics of turbomachinery flow fields. HIREP provides a facility in which the effects of rotor-tip/end-wall clearance, blade shape, and hydrodynamic loads on rotor efficiency, cavitation, wake effects, and dynamic response can be quantified.

ACKNOWLEDGEMENTS

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REFERENCES

1. Reeder, John A., "Tip Clearance Problems in Axial Compressors (A Survey of Available Literature)," Union Carbide Corporation, January 4, 1969.

2. Peacock, R. E., "Blade Tip Gap Effects in Turbomachines, A Review," Naval Post-Graduate School Report NPS 67-81-016, 1981.

3. Lakshminarayana, B., "Leakage and Secondary Flows in an Axial Compressor Cascade," Ph.D. Thesis, Liverpool University, September 1963.

4. Lakshminarayana, B., "Methods of Predicting the Tip Clearance Effects in Axial Flow Turbomachinery," ASME Trans., J. Basic Engineering, September 1970.

5. Rains, D. A., "Tip Clearance Flows in Axial Flow Compressors and Pumps," Hydromechanics and Mechanical Engineering Laboratories, California Institute of Technology, 1954.

6. Mitchell, A. B., "An Experimental Investigation of Cavitation Inception in the Rotor Blade Tip Region of an Axial Flow Pump," Admirality Research Laboratory, ARL/RI/G/HY/11/2, August 1958.

7. Gearhart, W. S., "Tip Clearance Cavitation in Shrouded Underwater Propulsors," AIAA J. Aircraft, Vol. 3, No. 2, March-April 1966.

8. Shuba, B. H., "An Investigation of Tip-wall Vortex Cavitation in an Axial-Flow Pump," Master's Thesis, The Pennsylvania State University, Department of Aerospace Engineering, 1983.

9. McBride, M. W., "Engineering Design Method for Axial and Mixed Flow Hydraulic Machinery," ASME Small Hydro Power Fluid Machinery 1982, pp. 73-80.

TABLE NO. 1

Table 1: Comparison of Theoretical and Experimental Performance Parameters

	Flow Coefficient	Pump Head Coefficient	Pump Rotor Efficiency	Torque Coefficient	Power Ratio
Predicted Values Mass averaged (V_{ref} = 15.2 m/s)	1.20	1.48	0.90	0.24	0.71
Experimental Values (V_{ref} = 10.7 m/s)	1.24	1.59	0.93	0.26	0.77
	1.31	1.42	0.91	0.22	0.63
	1.39	1.24	0.88	0.19	0.51
	1.44	1.11	0.86	0.17	0.43

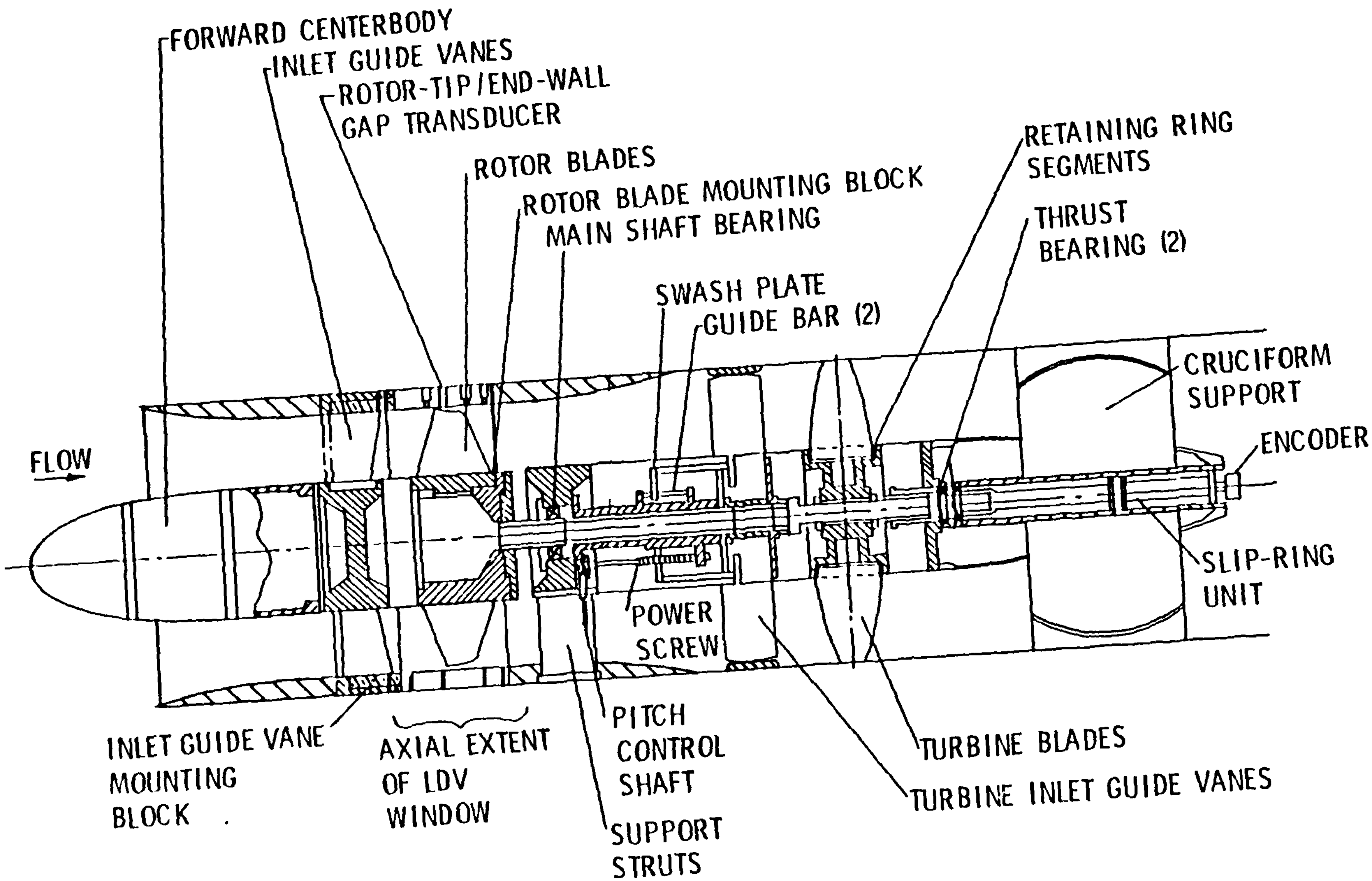

Figure 1. General Layout of the HIREP Facility.

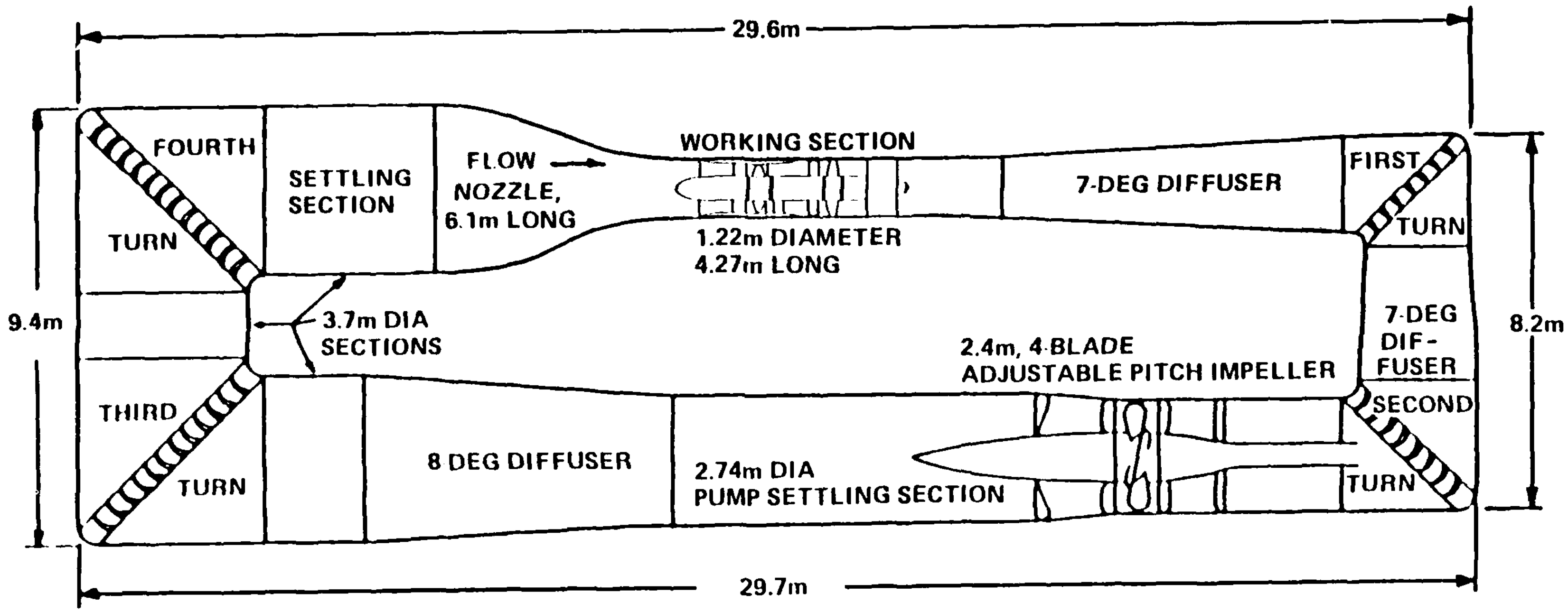

Figure 2. Installation of HIREP in the Garfield Thomas Water Tunnel.

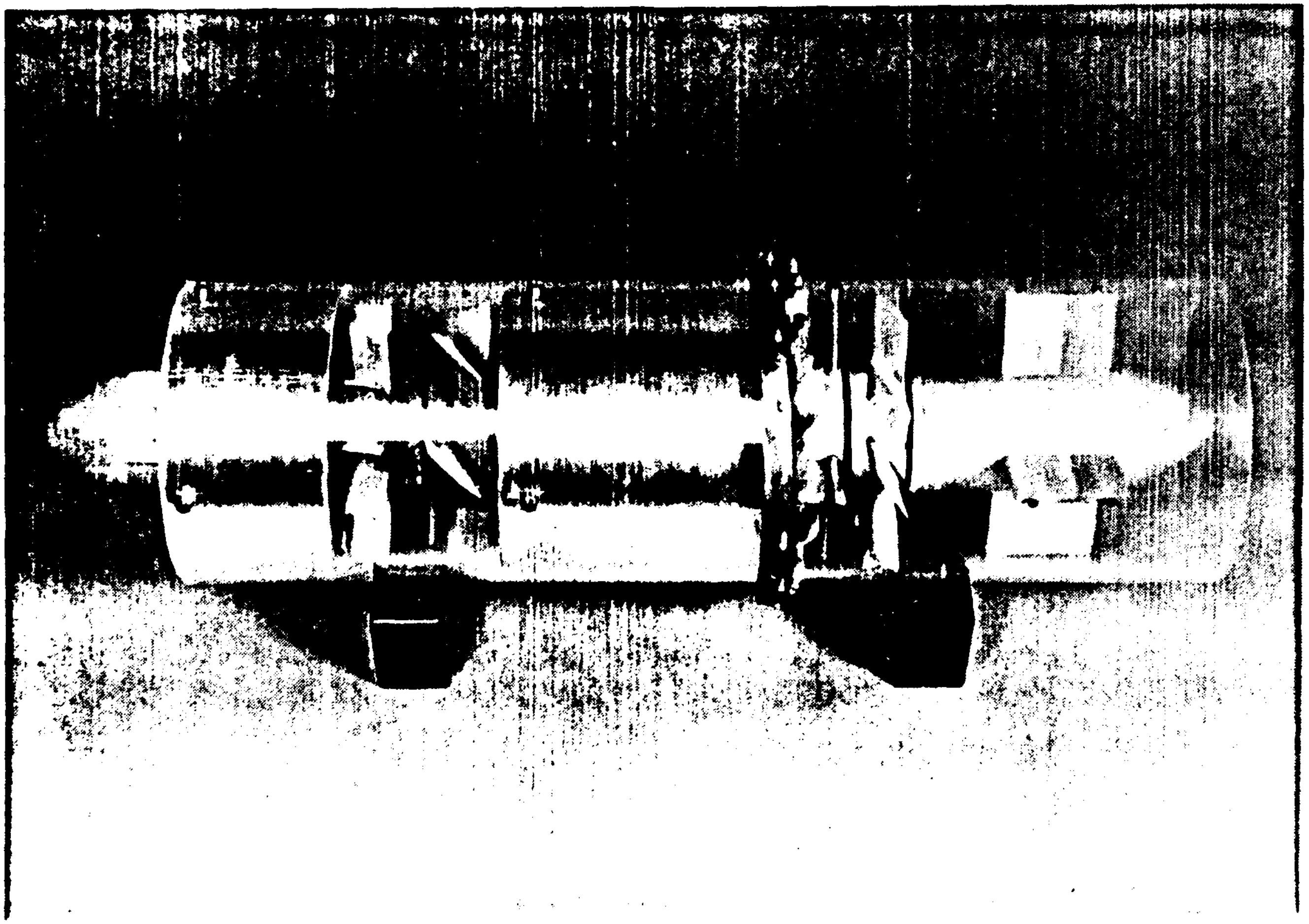

Figure 3. Photograph of HIREP One-sixth Scale Model.

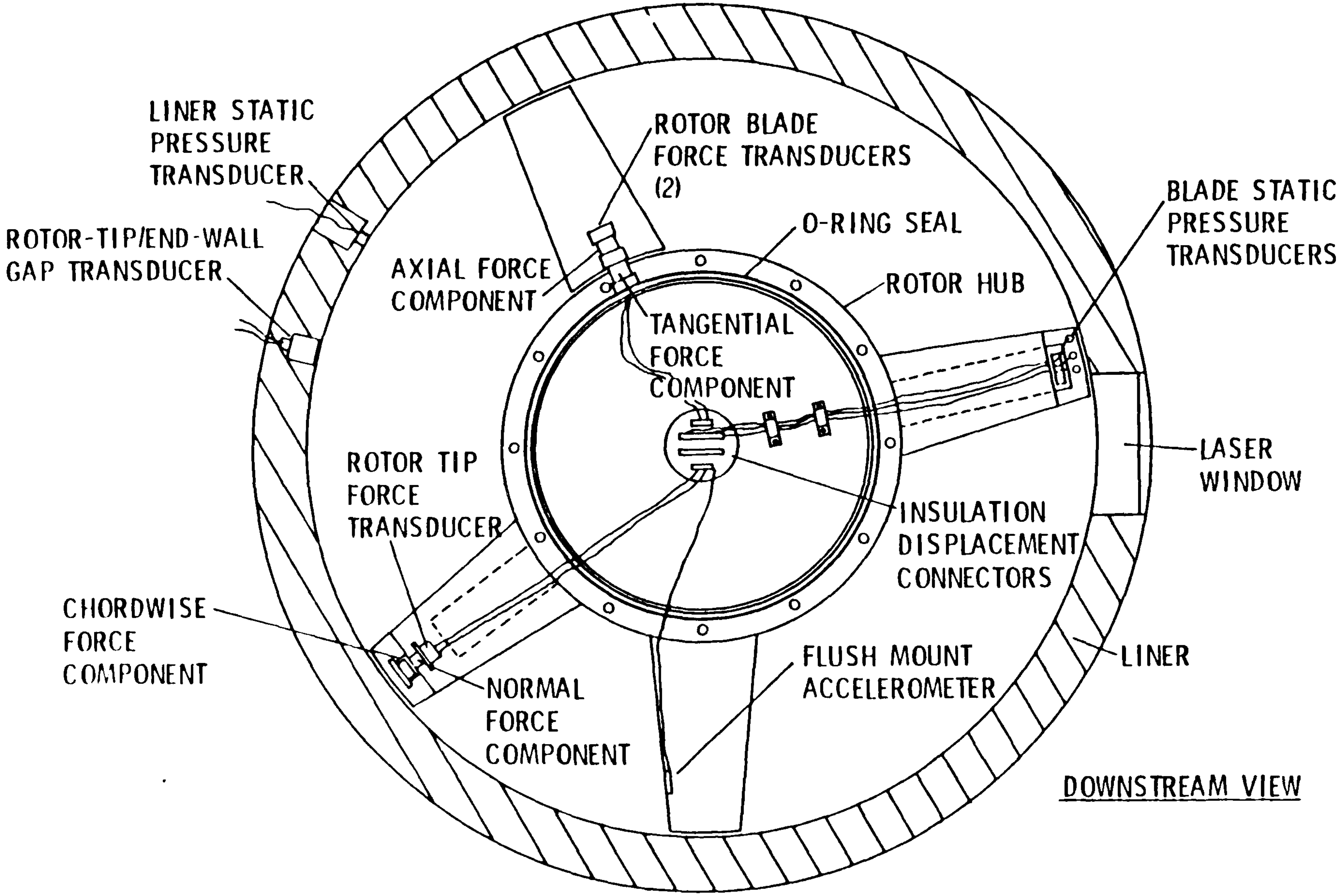

Figure 4. Rotor Hub Showing Instrumentation Capabilities.

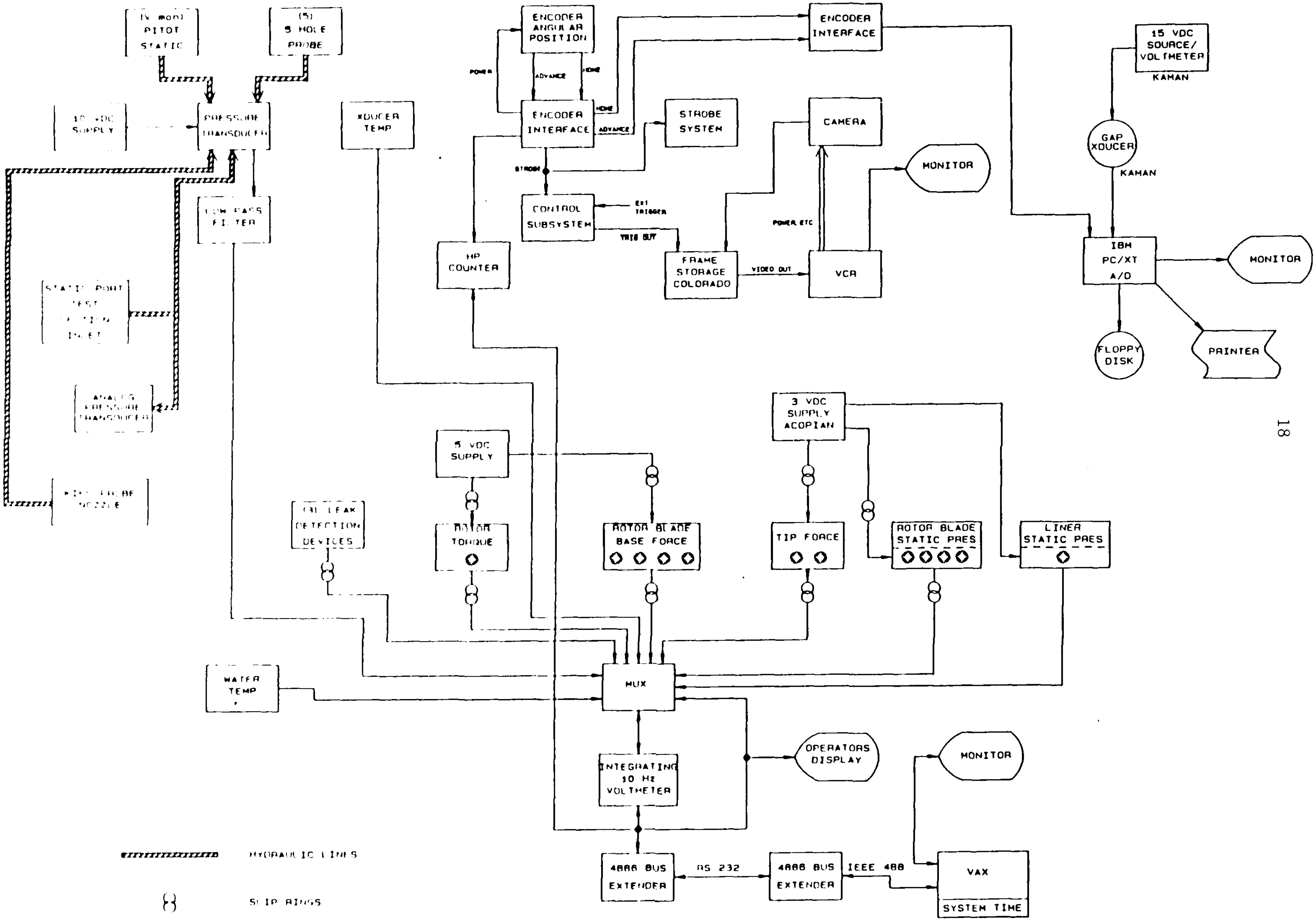

Figure 5. Instrumentation Block Diagram.

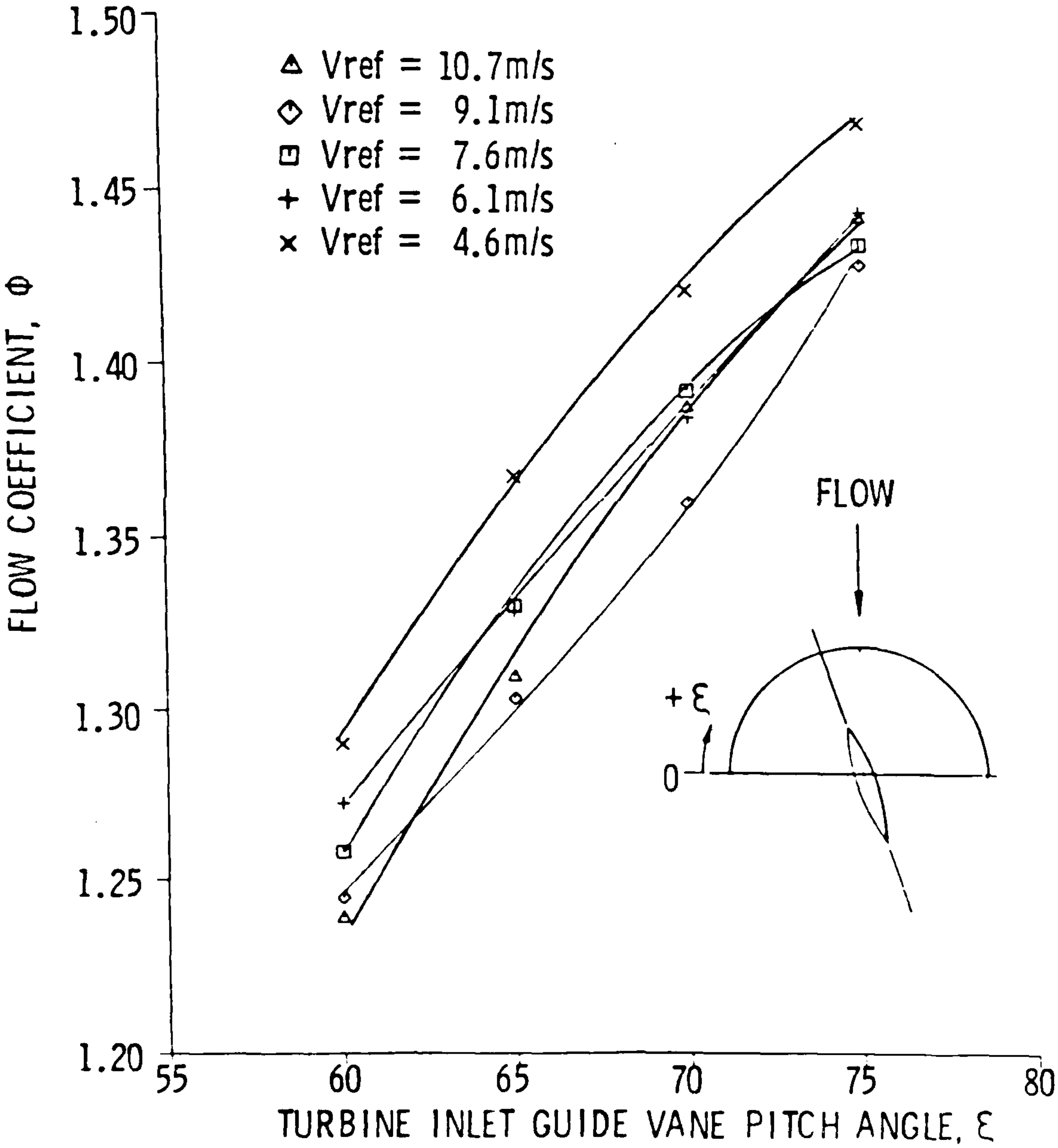

Figure 6. Flow Coefficient versus Turbine Inlet Guide Vane Pitch Angle.

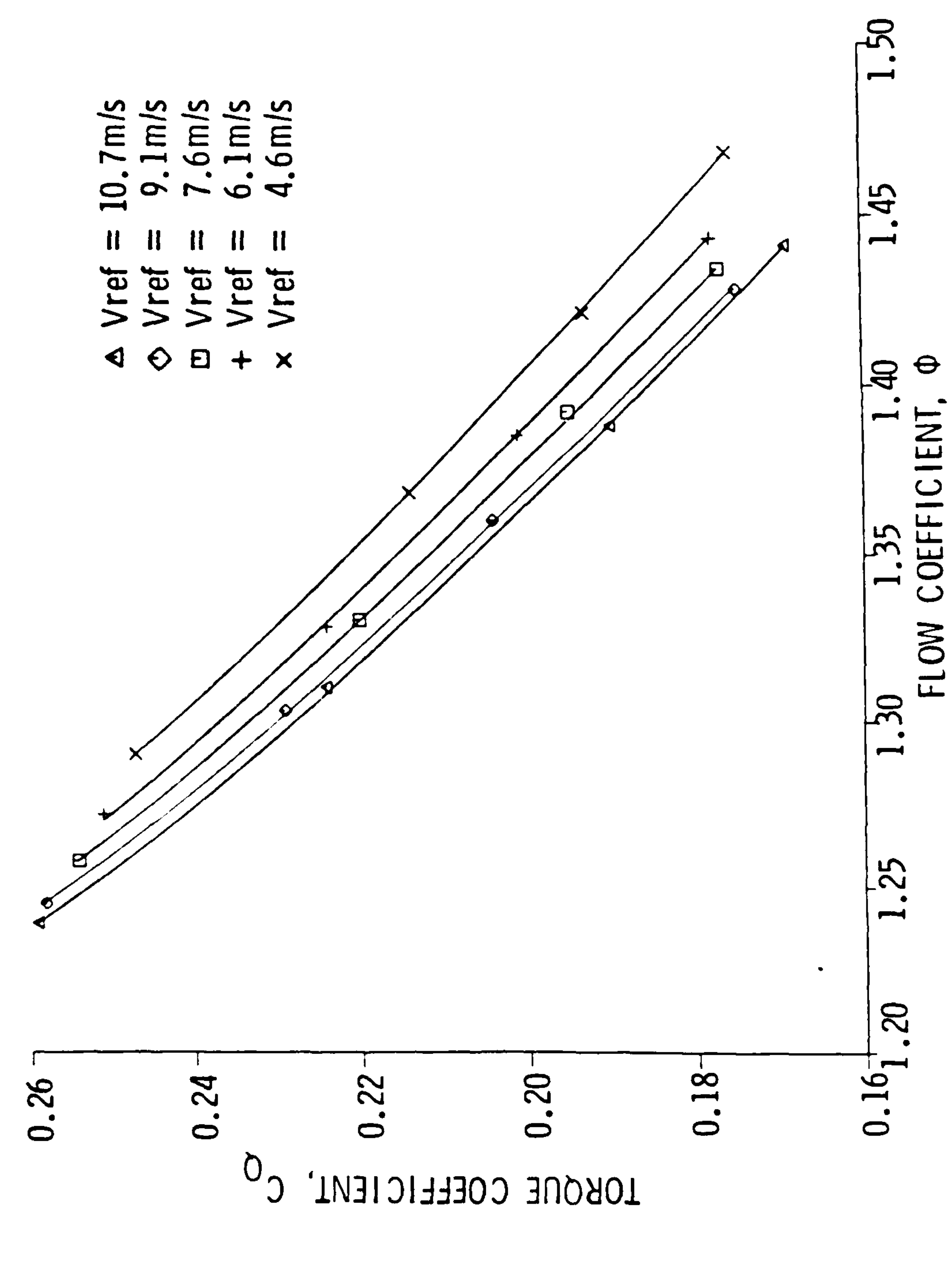

Figure 7. Torque Coefficient Versus Flow Coefficient.

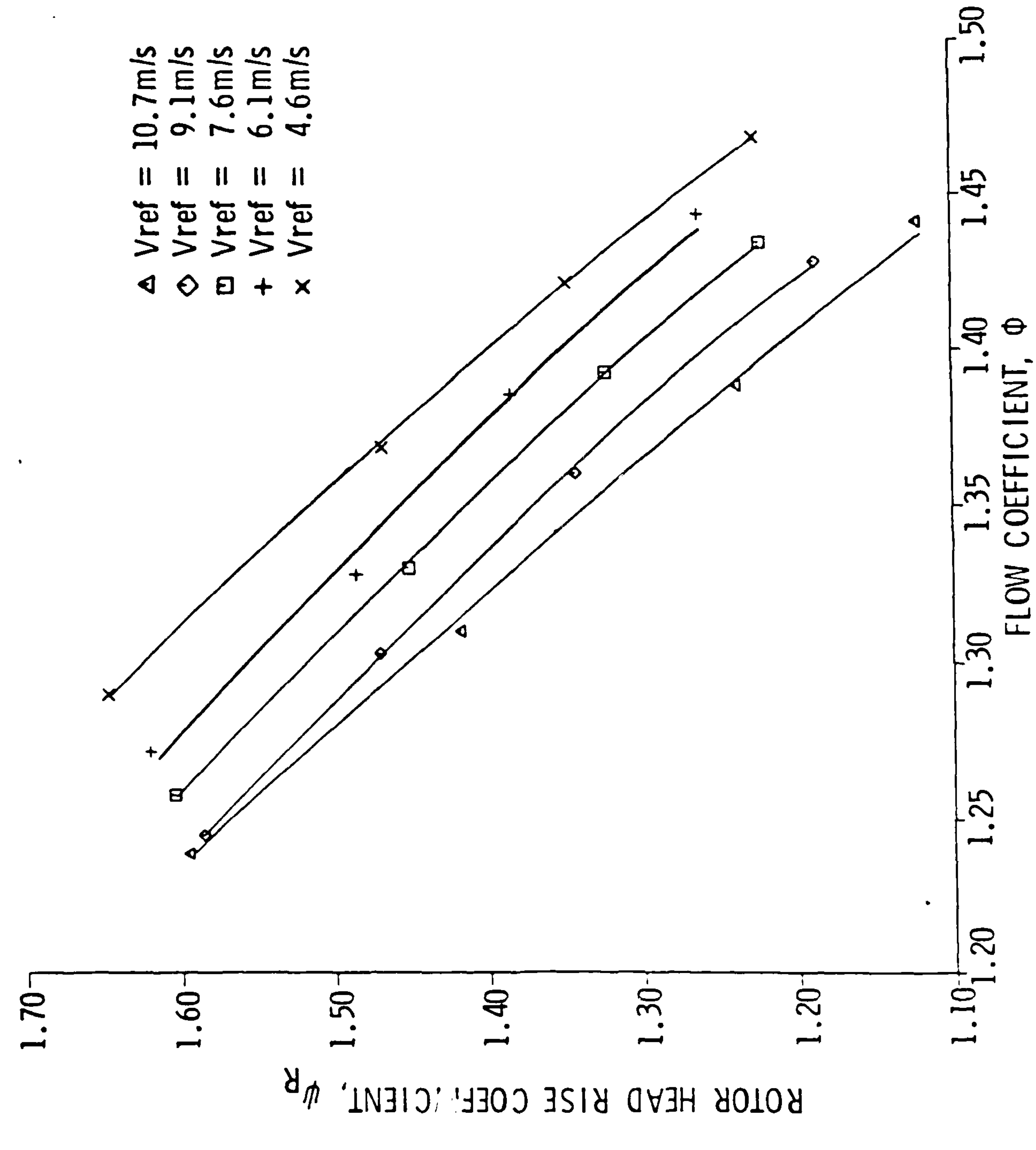

Figure 8. Rotor Head Rise Coefficient Versus Flow Coefficient.

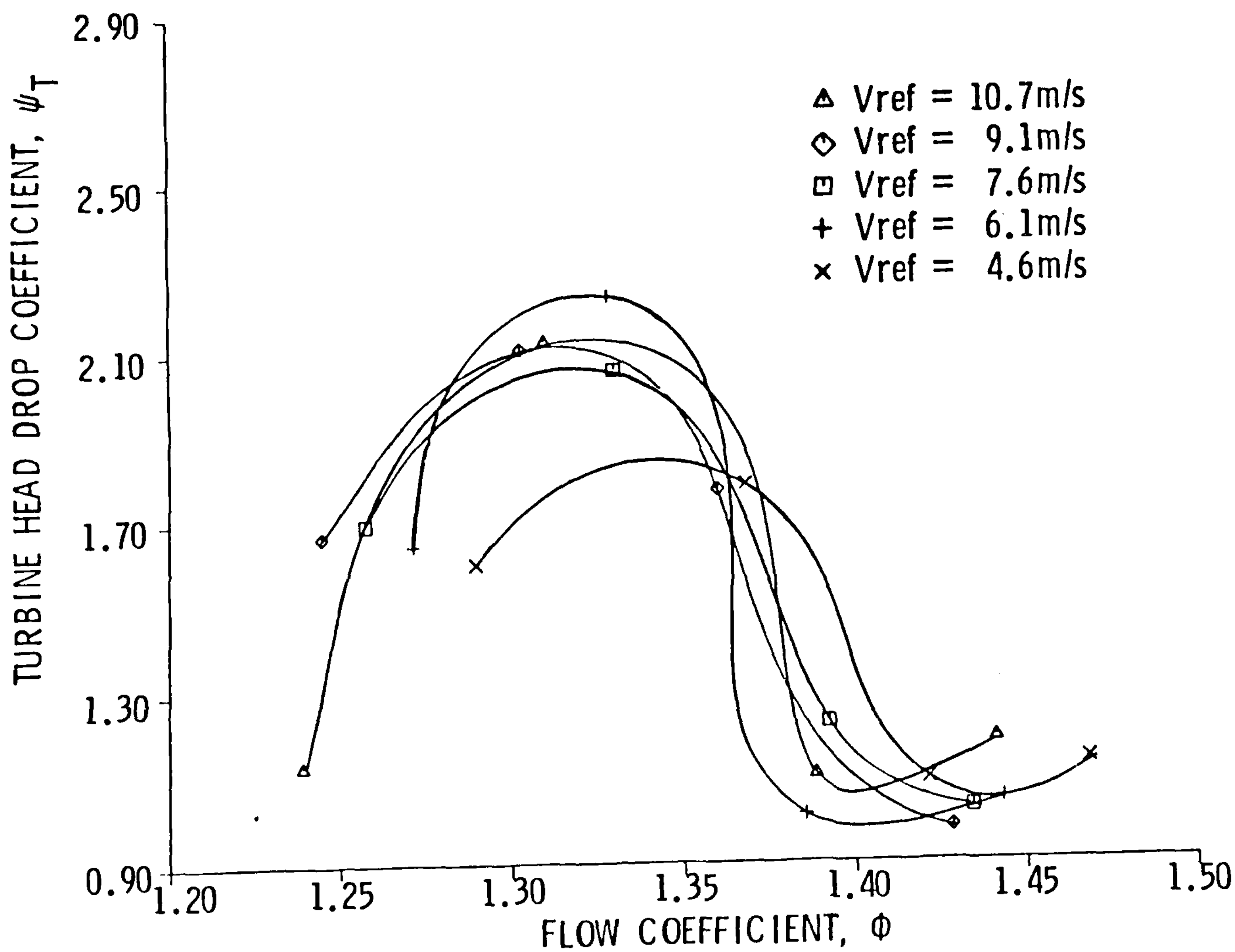

Figure 9. Turbine Head Drop Coefficient Versus Flow.

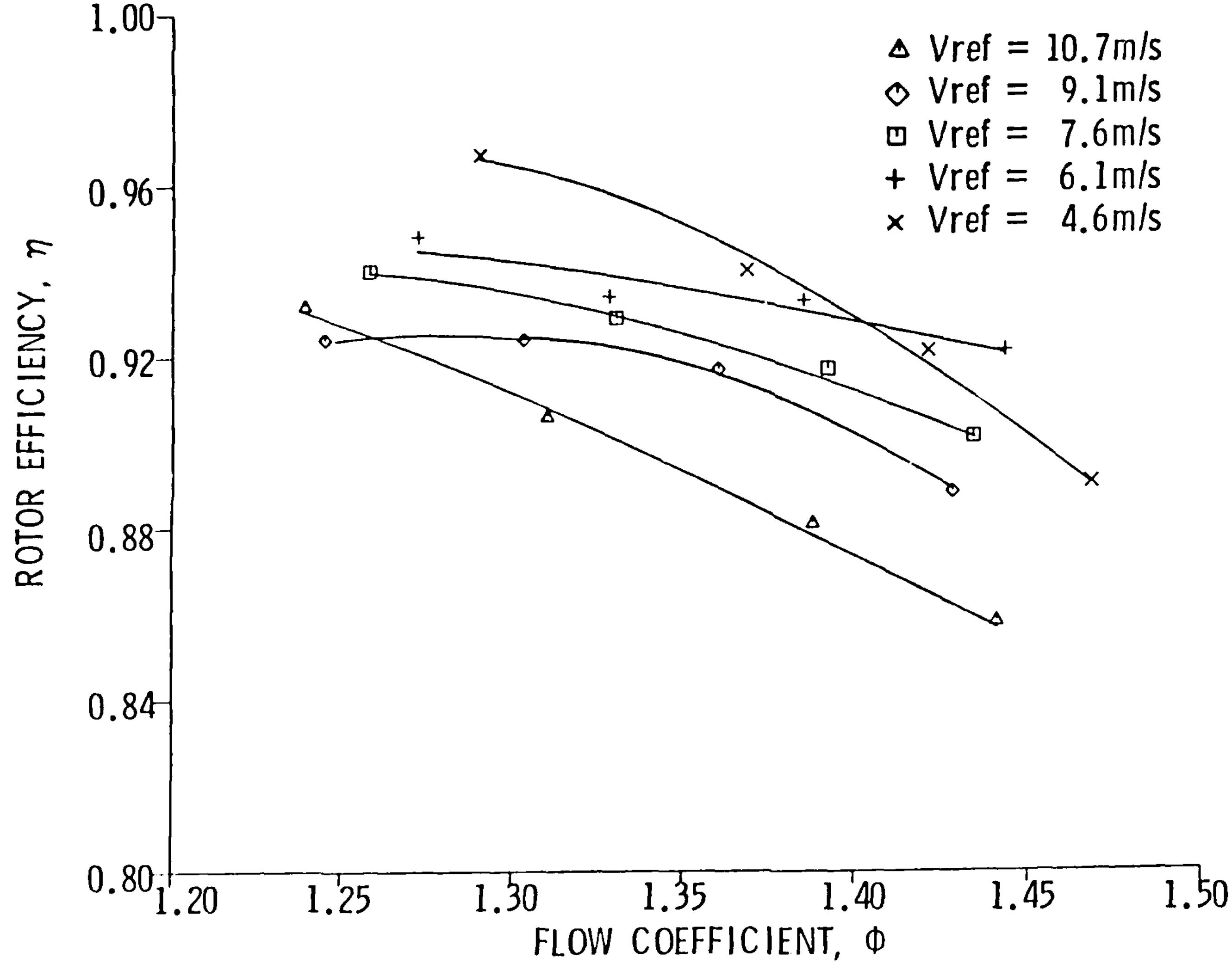

Figure 10. Rotor Efficiency Versus Flow Coefficient.

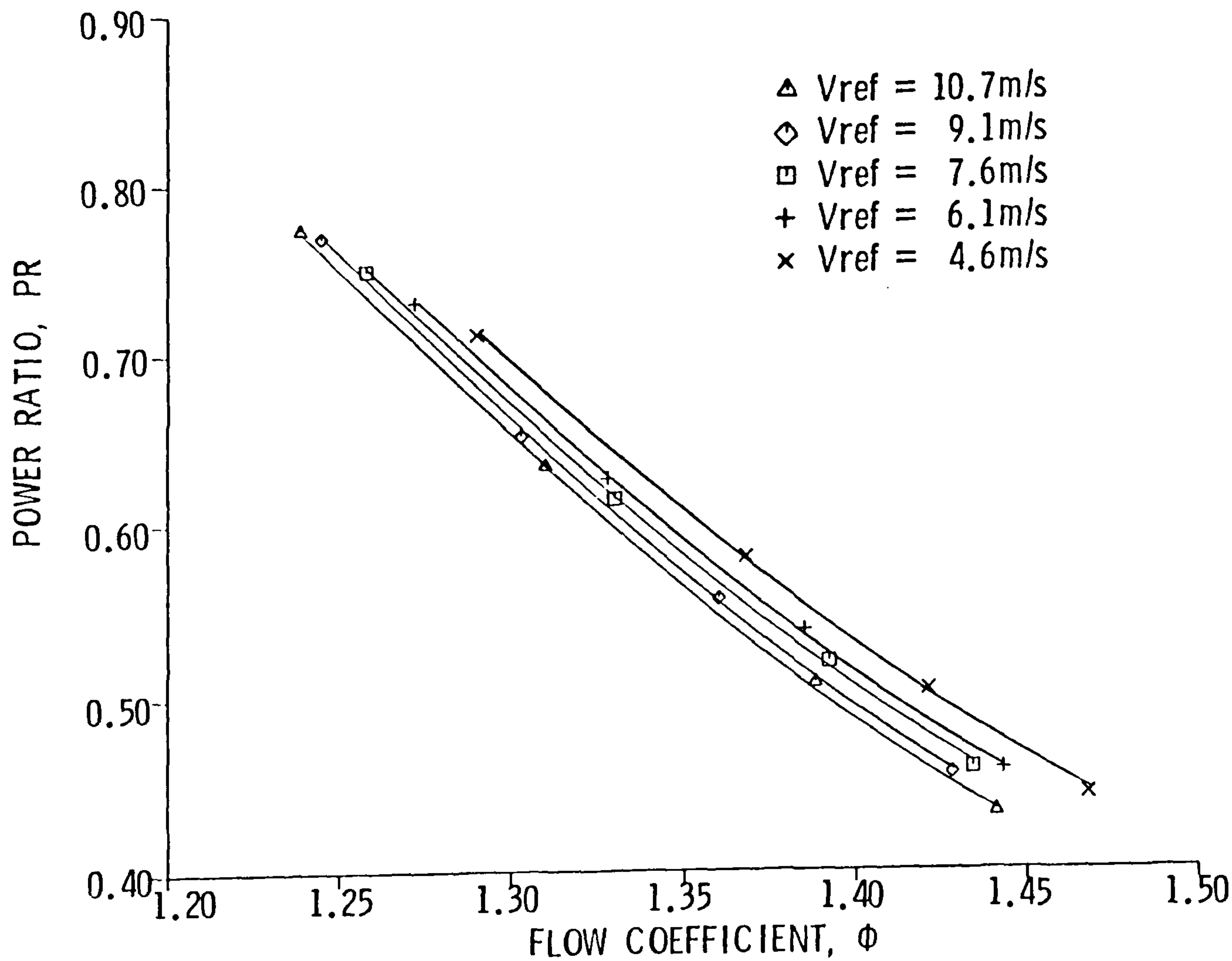

Figure 11. Power Ratio Versus Flow Coefficient.

Figure 12. Photograph of Leakage Vortex Cavitation.

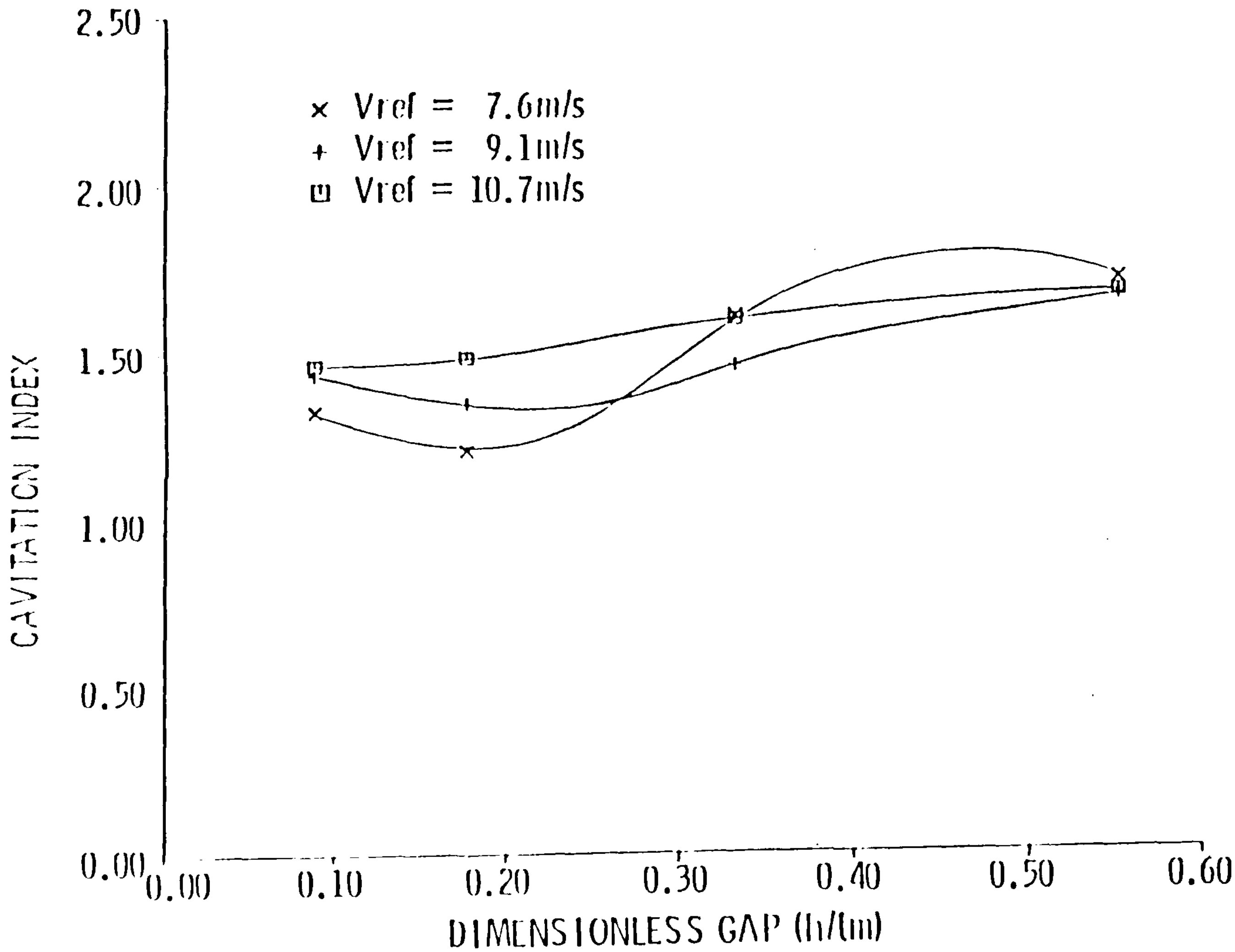

Figure 13. Cavitation Indices for the Leakage Vortex.

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